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Review of combustion characteristics of CI engines fueled with biodiesel

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ABSTRACT

In this study, the combustion characteristics of CI engine with biodiesel–diesel blends and neat biodiesel are critically reviewed. Combustion parameters like cylinder pressure, peak pressure, rate of pressure rise, start of combustion, ignition delay period, combustion duration, mass fraction burned, instantaneous heat release rate and cumulative heat release rate of biodiesel blends and diesel have been reviewed. The difference in in-cylinder peak pressure between diesel and biodiesel blends is not significant and is within 1%. The ignition of the biodiesel is earlier than that of diesel by about 1–2° CA. Both diesel and biodiesel fuels experience rapid premixed burning followed by diffusion combustion as it is typical for naturally aspirated engines. The instantaneous and cumulative heat release rates of both fuels are quite close to each other. In general, the combustion characteristics of biodiesel fuel blends and neat biodiesel have resulted in the same characteristics as for normal diesel combustion. The results reviewed in this article indicate that biodiesel fuel blends and neat biodiesel can be used as an alternative and environment friendly fuels without any major modification of the CI engines.

Contents

Abbreviations: CA, crank angle; CI, compression ignition; CO, carbon monoxide; DI, direct injection; CHR, cumulative heat release; HC, hydrocarbon; HRR, heat release rate; IDI, indirect injection; IMEP, indicated mean effective pressure; JME, jatropha methyl ester; PM, particulate matter; PME, palm methyl ester; PAH, poly aromatic hydrocarbon; RME, rapeseed methyl ester; SFC, specific fuel consumption; SME, soybean methyl ester; NO_x, nitrogen oxides; NHRR, net heat release rate

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1. Introduction

There is an increasing attention on biodiesel because it is a sulfur-free, non-toxic, biodegradable, oxygenated and renewable fuel. Many studies [1–5] have shown that the physical and chemical properties of biodiesel are very close to diesel fuel. Biodiesel has higher cetane number than diesel fuel, has no aromatics, almost no sulfur, and contains 10–11% oxygen by weight. These characteristics of the fuel reduce the emissions of carbon monoxide (CO), hydrocarbon (HC), and particulate matter (PM) in the exhaust gas compared to diesel fuel [6–8]. There are several reasons for biodiesel to be considered by both developing and industrialized countries because of energy security, environmental concerns, foreign exchange savings, and socioeconomic issues related to the rural sector.

A number of researchers [9,10] have shown that biodiesel has fuel properties and provides engine performance that is similar to that of diesel fuel. To improve air quality, severe emission regulations have been applied to diesel engines. The severe emission regulations in the world have placed design limitations on heavy-duty diesel engines. The trend towards cleaner burning fuel is growing worldwide. Recent studies [11,12] indicate that the cetane number, aromatic content and type, sulfur content, distillation temperature, and density are important factors for emission control. Because of the absence of sulfur and the presence of fuel oxygen, biofuels are considered very promising to reduce pollutants. Predicted depletion of fossil fuels in the foreseeable future and environmental pollution concerns provide motivation for the search of renewable alternative fuels, which would have relatively lesser harmful impact on the environment. The objective of this paper is to exclusively study the combustion characteristics of biodiesel fuel blends and neat biodiesel and to find the suitability of using them in CI engines.

2. Performance of biodiesel–diesel blends

Krahl et al. [13] reviewed the utilization of rapeseed oil, rapeseed methyl ester (RME) and diesel fuel in terms of exhaust gas emission and were of the opinion that stress should be given to RME than rapeseed oil alone. Rao and Gopalakrishnan [14] reported the results of tests conducted on DI engine with karanja oil, rice bran oil, palm oil and also with their corresponding methyl esters and they have recommended the use of methyl esters of vegetable oils rather than straight vegetable oils.

Rakopoulos et al. [15] conducted an experimental study with olive oil in DI and IDI engines. They used 25:75 and 50:50 blends of olive oil and diesel fuel and observed slightly increased SFC, unaltered maximum pressures and moderate increase in exhaust smoke. Agarwal and Das [16] studied the performance of a single cylinder DI diesel engine with methyl esters of linseed oil and found that 20% blend of biodiesel and diesel gave the best performance among all blends. Rosca et al. [17] studied the fuel injection characteristics of a biodiesel type fuel from waste cooking oil and concluded that it is possible to use B50 or B100 type fuels for fueling the diesel engine, if proper adjustments of the injection timing are made.

Senthil Kumar et al. [18] used JME as pilot fuel and orange oil as the inducted fuel in dual fuel diesel engine and observed reduced smoke levels and improved thermal efficiency. Rao and Mohan [19] studied the performance evaluation of DI and IDI engines with JME. They concluded that biodiesel operation with supercharging was the best technique for DI engines and biodiesel operation under naturally aspirated condition yielded the best results in the case of IDI engines. Sharp et al. [20] observed that 20% soy methyl ester-80% diesel blend (B20) reduced HC emission and there were small reductions in aldehydes and PAH.

Suryawanshi and Deshpande [21] tested pungamia methyl ester in CI engine and suggested considering timing changes, exhaust gas recirculation and variable geometry turbo charging to get better performance from biodiesel fuels. Sirman et al. [22] tested B20 soybean biodiesel in an advanced automotive diesel engine and observed reduced PM emissions, HC, CO and NO_x emissions. The authors suggested adjustment of engine parameters such as injection timing and exhaust gas recirculation levels for further reductions in emissions.

Majority of the studies on biodiesel emission characteristics have reported increase in NO_x emissions with biodiesel [23]. Advancing the combustion phasing [24], higher combustion temperatures [25], oxygen content of biodiesel [26] and differences in the chemical composition of diesel and biodiesel [27] are thought to be the possible causes of these effects of biodiesel on NO_x emissions. For a possible solution to the increase in the NO_x emission, new, modified injection strategies with multiple injection events have to be considered [28].

3. Performance of neat biodiesel

B100 or 100% biodiesel is seriously being considered as a fuel of choice by operators fueling in environmentally sensitive areas [29,30]. Scholl and Sorenson [31], Schumacher et al. [32] and Reece and Peterson [33] reported reductions in smoke density when fueling with biodiesel as compared to diesel. Reece used rapeseed derived biodiesel while Schumacher used soybean derived biodiesel fuels. Rao and Gopalakrishnan [14] however, noted increases in smoke density when they operated with pure karanja based biodiesel. Scholl and Sorenson [31] noted reductions in HC and CO with soybean derived biodiesel. Reece and Peterson [33] observed reductions in power ranging from 1% to 7% while Schumacher et al. [32] observed increased power of 3% using a Cummins 5.9 L DI turbocharged engine. Increased power was also observed by Feldman and Peterson [34] during a 200 h test using a 3 cylinder DI, naturally aspirated diesel engine with the injection timing advanced by 2°. Shifting the timing appears to be an appropriate and acceptable method that can be used to optimize the CI engine for biodiesel fueling.

Sharp et al. [20] tested a heavy-duty diesel engine with 100% soy biodiesel and observed a 10% increase in NO_x, a 77% reduction in PM, and a 25% reduction in CO. Schumacher et al. [32] observed that fueling of CI engines on 100% soybean methyl ester (SME) slightly reduced the power when compared to engines fueled with petroleum diesel fuel. The specific power developed by CI engine fueled on 100% biodiesel will vary depending on engine design and fuel delivery. CO, HC, PM and smoke exhaust emissions tend to be lower when fueled with biodiesel, while NO_x exhaust emissions tend to be higher. Identical results were observed by Sivalakshmi and Balusamy [35] when they have tested B100 prepared from neem oil. Materials from engine wear were found to be lower in the analysis of the engine lubricating oil (Fe, Pb, and Si).

4. Engine teardown and durability

Many researchers [36,37] have studied how vegetable oil based fuels will affect the engine's operational lifetime and subsequent emissions. They have reported problems with certain engine components and fluids like engine oil breakdown, oil dilution, and rubber degradation. Perkins et al. [36] ran a 1000 h test on three Yanmar engines fueled with 100% RME, 50% RME blended with diesel, and 100% diesel fuel. They found that the engines

fueled with RME performed similar to those with diesel with respect to engine wear and long term performance. However, a slight decrease in engine oil viscosity was experienced with the ester fuel. Ishi and Takeuchi [37] conducted 50 h of endurance experiments with transesterified jatropha curcas oil on a single cylinder pre-chamber type IDI engine and observed significant reduction in black smoke, CO and HC concentrations in exhaust.

Schumacher et al. [32] tested two Dodge pickup trucks fueled with 100% SME for a run of about 80,500 km each. Engine coking did not appear abnormal when inspected and engine oil analysis of the biodiesel fueled trucks showed lower engine wear compared to those with diesel in three key contaminants, Fe, Pb and Si. The 100% SME fuel rapidly deteriorated OEM rubber fuel lines. Fraer et al. [38] concluded that there was no injector replacement and no filter plugging or sludge accumulation in biodiesel-fueled engine and the maintenance cost of vehicles was the same as that of diesel vehicles.

5. Combustion characteristics of biodiesel

Combustion of fuels is one of the most important processes which affect the performance and emission characteristics as well as the engine durability [39]. The important parameters that signify the combustion process effectiveness are in-cylinder pressure, ignition delay, combustion duration, heat release and cumulative heat release rate [40,41]. In-cylinder pressure can be measured directly from the engine and the other combustion parameters can be calculated from the in-cylinder pressure. The heat release rate is estimated from the first law of thermodynamics using the in-cylinder pressure and the geometry of crank and connecting rod [39]. The other important combustion parameters can be easily estimated from the heat release rate variation over an engine cycle.

5.1. In-cylinder pressure

The in-cylinder pressure measurement is considered to be a very valuable source of information during the development and calibration stages of the engine. The in-cylinder pressure signal can provide vital information such as peak pressure, *P–V* diagram, indicated mean effective pressure, fuel supply effective pressure, heat release rate, combustion duration, ignition delay and so on [42]. Moreover, based on ideal gas and first law of thermodynamics it can be used in more complex calculations, for example, in air mass flow estimation, combustion diagnosis and NO_x prediction [43]. In a CI engine, the peak cylinder pressure depends on the burned fuel fraction during the premixed burning phase [35].

Sinha and Agarwal [44] examined the combustion characteristics in a DI transportation diesel engine running with diesel–rice bran oil methyl ester blend. A Mahindra and Mahindra made four cylinder DI diesel engine was used for measurement of combustion pressure, rate of pressure rise and other in-cylinder parameters such as rate of instantaneous heat release, cumulative heat release rate, mass fraction burned etc.

Cylinder pressure in a CI engine depends on the burned fuel fraction during the premixed burning phase, i.e., initial stage of combustion. Cylinder pressure characterizes the ability of the fuel to mix well with air and burn. High peak pressure and maximum rate of pressure rise correspond to the large amount of fuel burned in premixed combustion stage. The cylinder pressure crank angle history is obtained at different loads for diesel and B20. Peak pressure and maximum rate of pressure rise are obtained at different loads from these measurements.

Figs. 1 and 2 show the *p–θ* diagram for diesel and B20 fuels at 50% and 100% loads respectively. From these figures, it is clear that peak pressure increases as the load increases and for B20 fuel, combustion starts earlier compared to diesel fuel. Devan [45] investigated the combustion analysis of methyl ester–eucalyptus blends and diesel in a single cylinder four-stroke air-cooled Kirloskar engine. Fig.3 shows the variation of cylinder pressure with crank angle for standard diesel and methyl ester–eucalyptus oil blends. It is seen that high eucalyptus oil blends provide higher cylinder pressure compared to that of standard diesel. This is attributed to the lower cetane number of eucalyptus in the blend.

5.2. Peak pressure and rate of pressure rise

Figs. 4 and 5 show that peak pressure and rate of pressure rise are higher for B20 at low engine loads (up to 10% load) but become

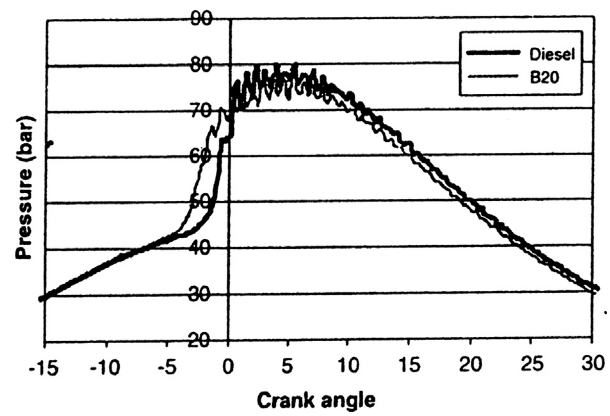

Fig. 1. *p–θ* diagram at 50% load, 1400 rpm for medium duty DI engine.

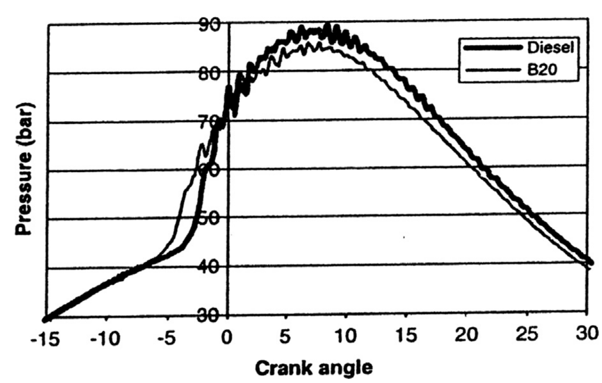

Fig. 2. *p–θ* diagram at 100% load, 1400 rpm for medium duty DI engine.

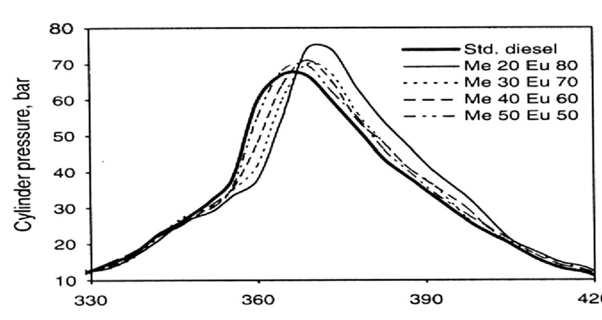

Fig. 3. Variation of cylinder pressure for methyl ester–eucalyptus blends at full load.

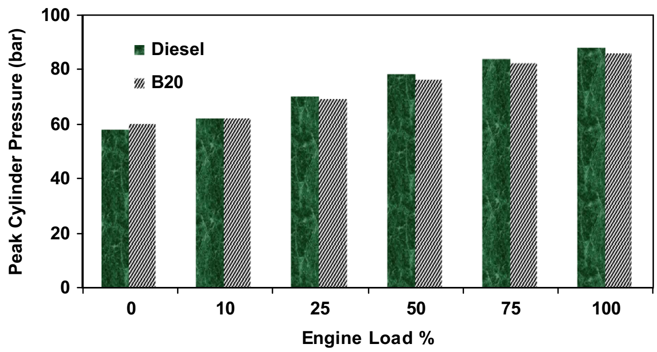

Fig. 4. Variation of peak cylinder pressure with engine load.

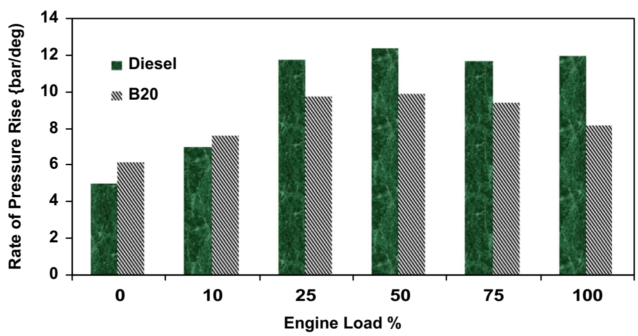

Fig. 5. Variation of rate of pressure rise with engine load.

lower when the engine load increases. A slight decrease of the peak cylinder pressure was observed by many researchers with the increase of biodiesel content in the blends [40,46,47]. However, the difference in in-cylinder peak pressure between diesel and biodiesel blends is not very much significant and it is within 1% [39].

The crank angle where the peak pressure occurs is shown in Fig. 6. It shows that maximum pressure occurs within the range of 2–7° CA after top dead center for diesel and B20 fuels at all loads. Pressure reaches its maximum later for B20 at higher loads which reconfirms that rate of pressure rise is lower at higher loads for B20.

5.3. Start of combustion

The start of combustion was calculated based on 5% of the heat release. It was also checked from the change in slope on the pressure–crank angle diagram suggested by Heywood [48]. Mustafa Canakci [1] studied the combustion characteristics of two different petroleum diesel fuels (No. 1 and No. 2) and biodiesel from soybean oil. The tests were performed at steady state conditions in a four-cylinder turbocharged DI diesel engine at full load at 1400-rpm engine speed. The start of injection is usually taken as the time when the injector needle lifts off its seat. The start of combustion is the cumulative effect of differences in the start of injection and changes in the ignition delay period. For B100, the start of combustion timing is earlier than that for No. 2 diesel fuel and this is confirmed by the HRR profiles shown in Fig.7. B100 fuel started to burn about 3.33° earlier than No. 2 diesel fuel while No. 1 diesel fuel started to burn about 0.25° latter than No. 2 diesel fuel. Tesfa et al. [39] observed that the ignition of the biodiesel fuel was earlier than the diesel fuel by 0.8°, 1°, 1.5° and 1.2° for 105 Nm, 210 Nm, 315 Nm and 420 Nm load conditions respectively. Gumus [40] also reported that the start of ignition for biodiesel was advanced by up to 1.65°.

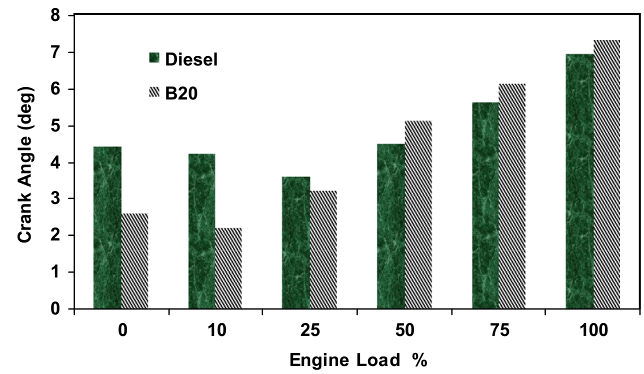

Fig. 6. Crank angle for peak cylinder pressure.

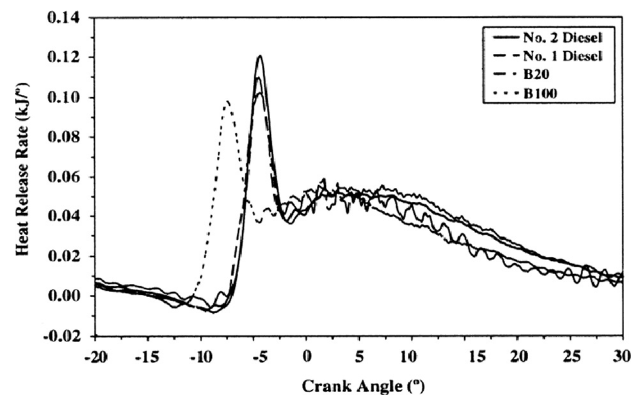

Fig. 7. Heat release rate for the fuels.

The advanced start of ignition is occurred due to the physical properties of biodiesel such as higher bulk modulus [49] higher viscosity and higher cetane number [50].

Qi et al. [51] reported that the combustion starts earlier for biodiesel and its blends than for diesel fuel at all engine loads. As the engine load is increased, the combustion starting point is closer for all the tested fuels. Ignition delay period decreases as the engine load increases because the gas temperature inside the cylinder is higher at high engine loads, which reduces the physical ignition delay period. The start of combustion may reflect the variation in ignition delay period because fuel pump and injector setting are kept identical for all tested fuels. In spite of the slightly higher viscosity and lower volatility of the biodiesel, the ignition delay period seems to be lower than that of diesel fuel. This may possibly be because a complex and rapid pre-flame chemical reaction takes place at high temperatures. As a result of the high cylinder temperature existing during fuel injection, biodiesel may undergo thermal cracking and hence lighter compounds are produced, which might have ignited earlier, resulting in a shorter ignition delay period [52].

5.4. Ignition delay

One of the most important parameters in the combustion phenomenon is the ignition delay. The ignition delay for any fuel can be calculated based on the duration between the start of fuel injection and start of combustion [53]. Ignition delay strongly depends on the type of fuel used and their concentrations in the cylinder charge. The associated changes in the charge temperature during compression, pre-ignition energy release, external heat transfer to the surroundings and the contribution of residual gases appear to be the main factors responsible for controlling the length of the ignition delay of the engine [54].

The physical and chemical properties of the fuels will affect the ignition delay period, and researchers have stressed that chemical properties are much more important than physical properties. The ignition quality of a fuel is usually characterized by its cetane number. Higher cetane number generally means shorter ignition delay. Mustafa Canakci [1] obtained the cetane numbers for No. 1 diesel fuel, No. 2 diesel fuel, and B100 as 45.3, 42.6 and 51.5, respectively. For B100 and No. 1 diesel fuel, the ignition delays were shorter than that for No. 2 diesel fuel. B100 and No. 1 diesel fuel had about 1.06°, 0.35° shorter ignition delay than No. 2 diesel fuel respectively.

Banapurmath et al. [53] investigated combustion characteristics of biodiesel from honge, jatropha and sesame oils on a four-stroke single cylinder direct injection CI engine. Fig. 8 shows the effect of brake power on ignition delay for diesel, methyl ester of Honge, Jatropha and Sesame oils, respectively, at optimum injection timings. They observed that ignition delay of biodiesel was longer than that of diesel, and the ignition delay decreased with the increase in brake power. Among the methyl esters tested, Sesame oil methyl ester (SOME) had shorter ignition delay compared to Honge oil methyl ester (HOME) and Jatropha oil methyl ester (JOME). Values of ignition delay were 11.5, 10.5 and 11 for JOME, SOME and HOME respectively, compared to 9.9° CA with diesel operation at 8% load.

Zhang et al. [55] investigated the combustion characteristics of turbocharged DI diesel engines using blends of methyl, isopropyl and winterized methyl ester of soybean oil with diesel as fuel. They found that all fuel blends except isopropyl ester had similar combustion behavior. Ignition delay for ester–diesel blend was shorter than that of diesel as a fuel. McDonald et al. [56] investigated soybean oil methyl ester as a fuel on a Caterpillar IDI diesel engine and found that overall combustion characteristics were quite similar to that of diesel except shorter ignition delay for soybean methyl ester. The increase in fuel viscosity, particularly for petroleum-derived fuels, results in poor atomization, slower mixing, increased penetration and reduced cone angle. These result in longer ignition delay for diesel fuel [46].

Scholl and Sorenson [31] investigated the combustion and experimental measurements of performance, emissions and rate of heat release for different fuel injection timings with soybean methyl ester in diesel engine. The initial combustion rates for the two fuels are almost identical. The diesel fuel has a slightly longer ignition delay, and also has a slightly higher maximum combustion rate during the premixed stage of combustion. There are no distinguishable differences in the combustion rate of the two fuels once the premixed combustion is completed.

Kinoshita et al. [57] investigated the performance and combustion characteristics of PME in a DI diesel engine and a swirl chamber diesel engine. They observed that PME showed shorter ignition delay than that of diesel fuel in both DI diesel engine and swirl chamber diesel engine. Identical result was observed by Orkun Ozener et al. [58] when they tested the combustion characteristics of soybean methyl ester. Shorter ignition delay was observed in the case of biodiesel and their blends [52]. The ignition delay was calculated by Ekrem Buyukkaya [46] for standard diesel, B5, B20, B70 and B100 fuels as 8.5°, 7.75°, 7.25°, 6.50° and 5.75° CA, respectively. Biodiesel usually includes a small percentage of diglycerides having higher boiling points than diesel. However, the chemical reactions during the injection of biodiesel at high temperature resulted in the breakdown of the high molecular weight esters. These complex chemical reactions led to the formation of gases of low molecular weight. Rapid gasification of this lighter oil in the fringe of the spray spreads out the jet, and thus volatile combustion compounds ignited earlier and reduced the delay period by about 3° CA for B100 compared to diesel fuel.

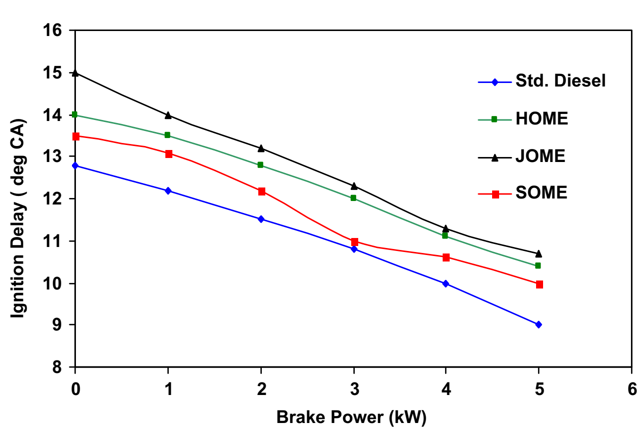

Fig. 8. Effect of brake power on ignition delay.

5.5. Heat release

The heat release rate diagram of the CI engines generally shows negligible heat release until toward the end of compression when a slight loss of heat during the delay period (which is due to heat transfer to the walls and to fuel vaporization and heating) is evident. Due to heat absorbed by the injected fuel from the cylinder, the heat release rate is slightly negative during the ignition delay period. The initial phase of combustion, called the premixed combustion, is very rapid because of the combustion of the fuel that has mixed with air during the ignition delay. After this phase, the combustion continues slowly until most of the fuel is burned. This phase of combustion is called mixing-controlled combustion.

The final combustion phase is the late or post combustion, which continues until the end of the expansion stroke. In this third stage a small but distinguishable rate of heat release persists throughout much of the expansion stroke. The heat release during this period usually amounts to about 20% of the total fuel energy [48]. All the fuel blends tested experience rapid premixed burning followed by diffusion combustion which is typical for naturally aspirated engines. After the ignition delay period, the premixed fuel–air mixture burns rapidly, releasing heat at a very rapid rate, after which diffusion combustion takes place, where the burning rate is controlled by the availability of combustible fuel–air mixture.

Grimaldi et al. [59] calculated the heat release rate for rapeseed methyl ester (RME) and they observed higher biodiesel combustion rate. They concluded that because of 12% lower heating value of RME than that of diesel, a higher mass is required to obtain the same energy release. The higher biodiesel burning rate is undoubtedly responsible for its higher NO_x emissions, although, on the other hand, it contributes to the reduction of soot formation.

Figs. 9 and 10 show the heat release rate diagrams for B20 and diesel fuels at half and full engine loads [44]. Both fuels experience rapid premixed burning followed by diffusion combustion as is typical for naturally aspirated engines. After the ignition delay period, the premixed fuel air mixture burns rapidly releasing heat at a very rapid rate, after which diffusion combustion takes place, where the burning rate is controlled by the availability of combustible fuel–air mixture. By analyzing these diagrams, it can be observed that when engine is fueled with B20, the combustion starts earlier under all operating conditions and also B20 shows shorter ignition delay compared to diesel fuel. The premixed combustion heat release is higher for diesel, which is responsible for higher peak pressure and higher rate of pressure rise.

Figs. 11 and 12 show the cumulative heat release for both fuels at different engine loads. These diagrams reconfirm the early onset of heat release for biodiesel blend. Cumulative heat release is also lower for biodiesel blend compared to diesel fuel possibly because of lower calorific value of biodiesel blend. The experimental investigation revealed that the overall combustion characteristics were quite similar for B20 and diesel fuel. However, combustion started earlier in the case of B20. Ignition delay was lower and combustion duration was slightly longer for B20 compared to diesel fuel. Lower heat release rates were found for B20 compared to diesel during premixed combustion phase. Total heat release was lower in the case of B20 in comparison to diesel fuel. 20% blend of RME did not cause any fuel/combustion related problems. Sukumar Puhan et al. [60] tested linseed oil methyl ester, jatropha oil methyl ester and coconut oil methyl ester in DI diesel engine and reported that the HRR and CHR for all the biodiesel fuels were almost similar. Similarly Xin Wang et al. [61] Marina Kousoulidou et al. [62] and Linxiao Yu et al. [63] reported that the HRR of biodiesel and diesel fuels were quite close to each other.

Senatore et al. [64] investigated the HRR of 100% RME and diesel fuel. Figs. 13 and 14 show the HRR diagram for two different operating conditions ($\Phi=3.75$ and $\Phi=1.61$) for pure diesel fuel and pure RME respectively. These figures point out that the HRR initially follows a downward trend, corresponding to the end of compression stroke which suddenly changes slope at the start of combustion. They found that the RME heat release took place in advance as compared to diesel and injection also started earlier in the case of RME as fuel and average cylinder gas temperature was also higher. It can be observed that when the engine is fueled with RME, the process starts in advance in all operating conditions, a feature which becomes more evident as the load increases. This determines a similar trend in the mean temperature variation rate of gases in the cylinder as shown in Figs. 15 and 16. Depending on

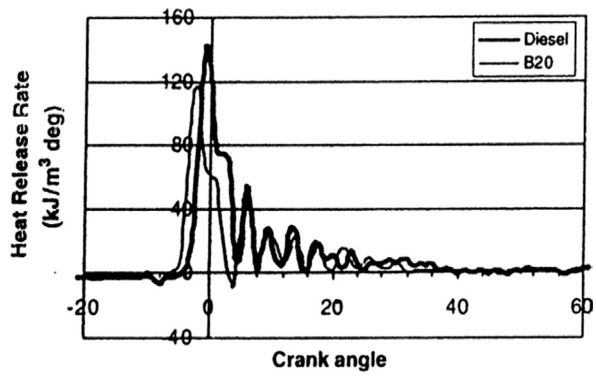

Fig. 9. HRR for 50% engine load for medium duty engine.

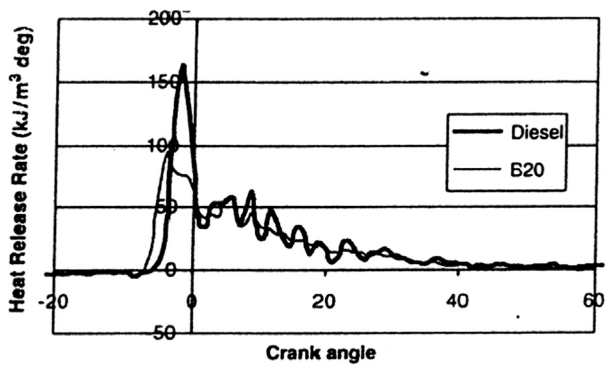

Fig. 10. HRR for 100% engine load for medium duty engine.

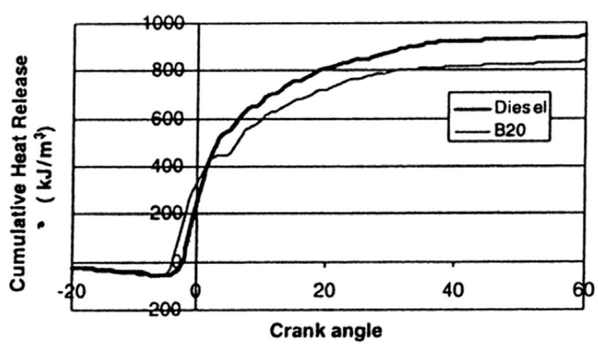

Fig. 11. CHR at no engine load for medium duty engine.

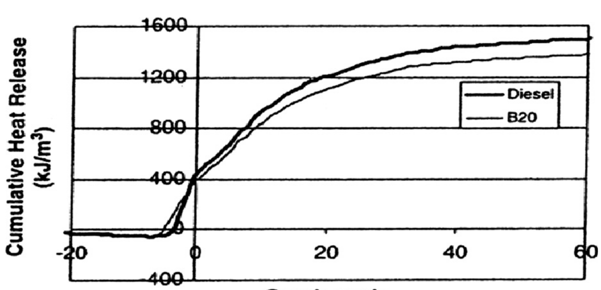

Fig. 12. CHR at 100% engine load for medium duty engine.

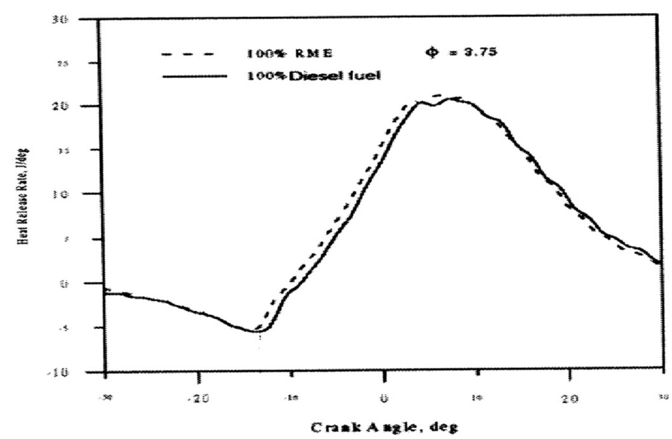

Fig. 13. HRR versus crank angle at $\Phi=3.75$.

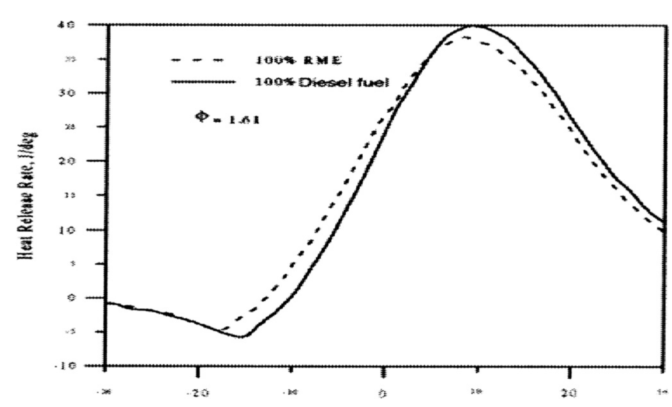

Fig. 14. HRR versus crank angle at $\Phi=1.61$.

the fuel used, the maximum temperature increase rate is found at different engine crank angle positions. When biodiesel is used, the temperature variation rate peaks at a position closer to piston top center.

Scholl and Sorenson [31] investigated the combustion of soybean methyl ester in a DI diesel engine and compared it to that of a conventional diesel fuel. The instantaneous rate of combustion was calculated in terms of mass burning rate divided by the total mass burned. They observed that the SME had a slightly shorter ignition delay and nearly identical cylinder pressure characteristics. The initial combustion rates for SME and diesel fuels were nearly identical. The diesel fuel had a slightly longer delay, and also had a slightly higher maximum combustion rate during the premixed stage of combustion. Once premixed combustion stage was over, there were no distinguishable differences in the combustion rate of the two fuels.

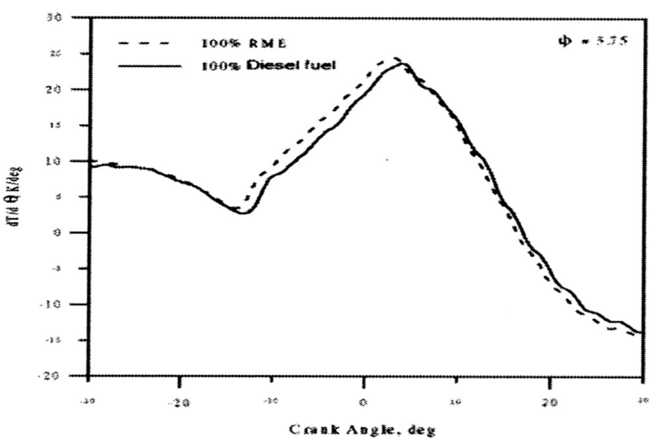

Fig. 15. Gases mean temperature variation rate versus crank angle at Φ=3.75.

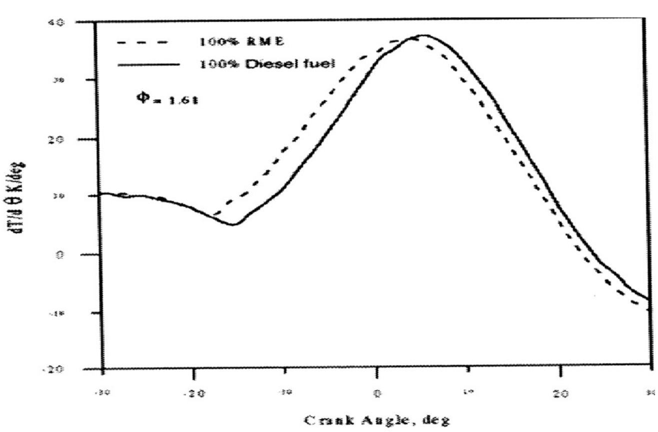

Fig. 16. Gases mean temperature variation rate versus crank angle at Φ=1.61.

5.6. Combustion duration

The combustion duration can be calculated based on the duration between the start of combustion and 90% CHR. Tsolaskis et al. [65] studied the engine performance and emissions of a diesel engine operated on diesel–RME blends. Fig. 17 shows the effect of fuel blend composition on the cylinder pressure and net heat release rate (NHRR) at IMEP of 4.5 bar and 6.1 bar. The authors observed that the increase of RME percentage in the fuel blend appears to reduce the ignition delay, increase the rate of fuel burnt in the premixed phase and shift the start of combustion to an early stage and hence increase the in-cylinder pressure compared to petroleum diesel combustion. Biodiesel such as RME is less compressible than diesel fuel, so the pressure in the pump-line-nozzle type fuel injection system can develop faster, and pressure waves can propagate faster in biodiesel than diesel even at the same nominal pump timing. As a result, the injection of biodiesel fuel starts earlier with higher pressure and rate and at the same CA degree, the mass of biodiesel injected is higher than the corresponding mass of diesel. The increased viscosity of RME leads to reduced fuel losses during the injection process, which leads to a faster evolution of pressure and, thus, advances the injection timing [43,66]. The combustion of the increased injection pressure and similar cetane number of RME compared to diesel, results in an increased amount of fuel undergoing premixed combustion at an early stage. The higher density of RME in conjunction with the increased injection pressure, results in delivery of a higher amount of fuel at the same injection setting conditions. Combustion, therefore, takes place over a shorter period of time, and this possibly allows less time for cooling by heat transfer and dilution, which results in higher NO_x formation associated with the combustion of RME.

Shorter combustion duration was observed for biodiesel by An et al. [47] and Bittle et al. [67]. Xin Wang et al. [61] studied the combustion characteristics of biodiesel fuel and compared with diesel in a heavy-duty turbocharged, common-rail diesel engine. The comparison of the start of combustion and combustion duration manifested that the start of combustion of diesel and biodiesel fueling was quite close in different operating conditions. Similarly, the combustion duration for the two fuels was very similar.

Atul Dhar and Avinash Kumar Agarwal [68] studied the combustion characteristics of karanja biodiesel in a transportation engine. They observed that all biodiesel blends showed shorter combustion duration for 1.4 or 2.8 bar BMEP at both engine speeds

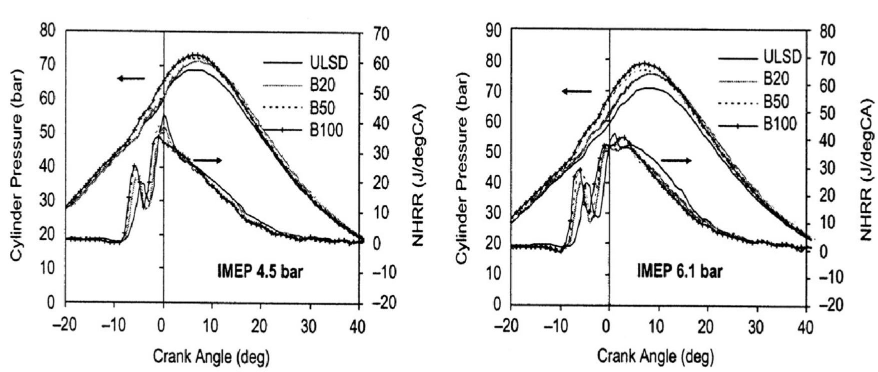

Fig. 17. Effects of fuel blend composition on the cylinder pressure and NHRR at IMEP 4.5 and 6.1 bar.

of 1800 and 2400 rpm. With increasing engine loads, cylinder temperature also increases, which increases the air–fuel mixing as well as vaporization of fuel. Quantity of fuel injected into the cylinder also increases with increasing engine load, which tends to increase the time required for completion of combustion of higher injected fuel quantity. Due to resultant effect of these two causes, combustion duration was found to be shorter at intermediate engine loads for all test fuels. Combustion duration for lower Karanja biodiesel blends was found to be lower compared to mineral diesel for all operating points. For higher biodiesel blends, combustion duration was shorter than mineral diesel at higher engine loads however, it was longer at lower engine loads. Anand et al. [69] also reported shorter combustion duration of Karanja biodiesel at higher BMEP and longer combustion duration at lower engine speeds and lower loads in comparison to mineral diesel.

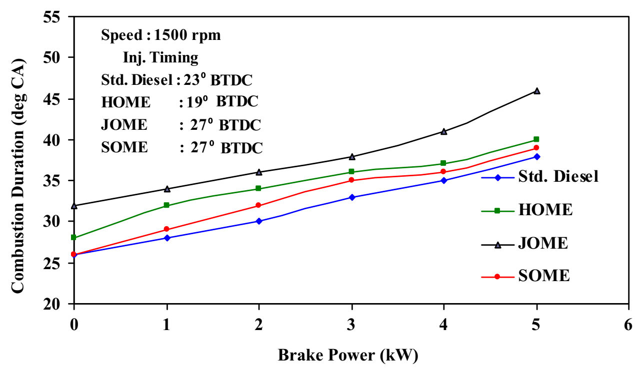

Fig. 18. Effect of brake power on combustion duration.

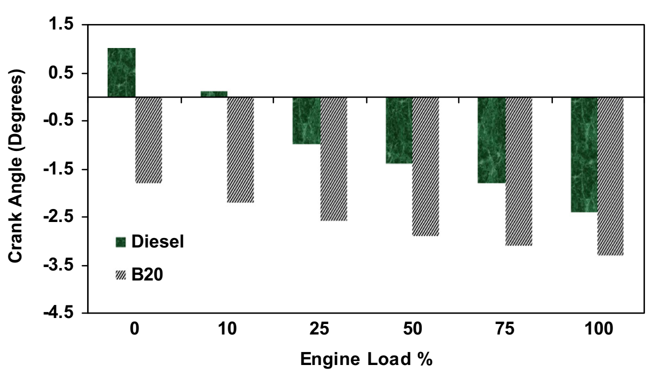

Fig. 19. Crank angle for 10% mass burn for medium duty engine.

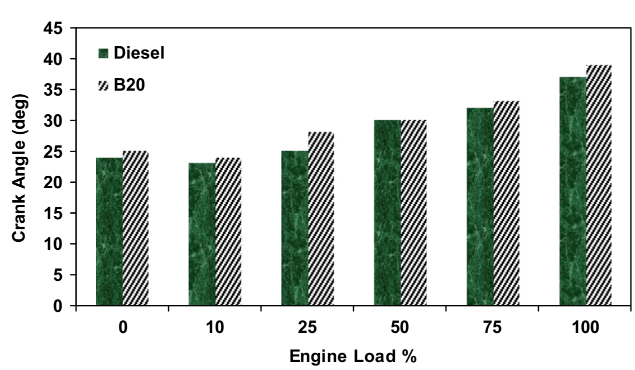

Fig. 20. Crank angle for 90% mass burn for medium duty engine.

At lower engine loads, longer combustion duration of biodiesel can be attributed to its slower evaporation characteristics because of its inferior fuel sprays having larger droplet sizes.

Fig. 18 shows the effect of brake power on the combustion duration [53]. The combustion duration increases with the increase in the quantity of fuel injected. Higher combustion duration is observed with all esters than diesel due to the longer diffusion combustion phase. This is contrary to the results obtained by Tsolakis et al. [65] and Atul Dhar and Avinash Kumar Agarwal [68].

5.7. *Mass burning rate*

Fig. 19 shows the crank angle at which 10% mass of fuel is burned. This figure shows that 10% of fuel burns earlier for biodiesel blend [44]. Fig. 20 shows the crank angle at which 90% mass of the fuel is burned. This figure shows that 90% of the fuel is burned earlier in the case of diesel, showing faster burning rate for diesel fuel. Increase in combustion duration is due to slow combustion of the injected fuel. Combustion duration for both the fuels increases as the load is increased due to increase in the quantity of fuel injected.

Gumus [40] studied the combustion and heat release characteristics of hazelnut kernel oil methyl ester–diesel blends in a DI compression ignition engine and observed that combustion starts earlier for biodiesel blends under all engine operating conditions and it becomes more prominent with higher biodiesel addition in the blends. The premixed combustion HR is higher for diesel owing to higher volatility and better mixing of diesel with air. Another reason may possibly be the longer ID of diesel, which leads to a larger amount of fuel accumulation in the combustion chamber at the time of the premixed combustion stage, leading to a higher mass burning rate. Mass burning rate decreased with the increase of biodiesel in the blends. He noted that the maximum burning rate of biodiesel was lower than that of diesel fuel owing to lower premixed burning of biodiesel. At the time of ignition, fuel air mixture prepared for combustion decreases with addition of biodiesel content in the blend due to lower volatility, ignition delay and higher viscosity. Therefore, the premixed heat release decreases with the addition of biodiesel content in the blend.

6. Conclusions

Combustion characteristics of CI engines fueled with neat biodiesel and biodiesel–diesel blends reported by various researchers are discussed. The main conclusions of this study are:

- It should be pointed out that several studies dealing with the emission, performance and combustion characteristics of biodiesel fuels did not reach unequivocal results and different behaviors have often been registered depending on the engine type, on the operating and maintenance conditions, on the testing methods, on the injection system and on its calibration, etc.
- However, the results obtained by the various researchers on biodiesel fuels show the performance and emission characteristics are similar to that of diesel fuel.
- The heat release rate of neat biodiesel and its blends with diesel follow the same trend as diesel fuel.
- In general, comparison of the combustion characteristics of biodiesel fuel blends and neat biodiesel has resulted in the same characteristics as observed for normal diesel combustion.
- The results reviewed in this article indicate that biodiesel fuel blends and neat biodiesel can be used as an alternative and environment friendly fuels without any major modification of the CI engines.

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